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HEAT TRANSFER THROUGH PHASE CHANGE AND CENTRIFUGATION APPLICATION TO TURBINE BLADE COOLING

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16. Abstract

Theoretical analysis of heat transfer conditions obtainable through phase change in a metallic heat carrier exposed to such strong centrifugation as that encountered in the case of turbine blades. The main concerns of the study are the kinetics of condensed droplets and the closed-cycle heat transfer performance through peripheral evaporation and condensation at the externally cooled hub. Following verification of the operating principle underlying the proposed heat transfer technique by means of and ethyl alcohol and cesium containing pressure vessel centrifugated at the end of a revolving arm, the requirements are examined for a high-temperature experiment at high acceleration values.

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APPLICATION TO TURBINE BLADE COOLING

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Summary

Heat transfer conditions obtainable by phase change in a metal heat carrier under the strong conditions of centrifugation encountered in the case of turbine blades are analyzed theoretically insofar as the the kinetics of the condensed droplets and the closed-cycle heat-exhange performances through peripheral vaporization and external cooling of the hub are concerned. HE2.2.1*

Following verification of the applicability of the principle of this transfer technique on a rotating arm providing an acceleration of the order of 10^3 ms^{-2} and carrying a chamber containing ethyl alcohol or cesium, the experimental modalities with sodium with high-velocity and high temperature testing means are examined.

1. Introduction

The advantages of heat transfer through phase change in a suitable heat carrier circulating in a closed circuit have been recognized in numerous cases of application. Producing slight mass displacements of a fluid selected on the basis of its vapor pressure and vaporization heat characteristics at the quasi-constant temperature of the cycle, heat sinks are currently used to extract intense heat fluxes, particularly with a suitable metal from the core of nuclear piles.

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*Number in the margin indicate pagination in the foreign text.

In every other field of industrial application, the cooling of gas turbine blades can be considered through phase changes of an alkaline metal previously introduced into the hollow blades with channels sealed at each end. Thanks to the great disparity of densities of the vapor and liquid phases, the heat transfer cycle can be maintained by centrifugation of the liquid phase provided external cooling of the base of the blades is secured.

Although the cooling of the blades by the thermosiphon cycle has been proposed by extension, the phase-change technique in high-speed rotors does not appear to have been the object of systematic studies. Resorting to small displacement only, this technique makes it possible, moreover, to elude the appearance of high stresses due to intensification of the hydrostatic pressure of the occluded fluid and to ensure virtually perfect isothermy of the blades, thus facilitating heat exchanges at their base.

The purpose of the study undertaken by ONERA is an initial verification of the workability of this process using a revolving arm device having low centrifugal acceleration and relatively low temperature ($\leq 10^3$ msec^{-2}), followed by an experiment at a high rotational velocity (acceleration up to $3 \cdot 10^5$ msec^{-2}) with sodium /HE2.2.2 as the transfer fluid, using testing means developed by the Atomic Division of SNECMA [1].

A theoretical analysis of the thermokinetic cycle provided, furthermore, for a sensible design of the geometry of the channels and of the exchangers that that extend the blades, as well as the conditions of fractional filling of the channels.

2. Heat Transfer Through Phase Change Cycle with Ce trifugation

The heat transfer process through phase change, designated as "evaporative-cycle thermosiphon" in the literature [2], takes advantage of the centrifugal forces imparted by the walls of the movable support, in order to increase the speed of circulation of the heat carrier. The prospects afforded by this device seem to have been recognized for the first time by H. Cohen and F. J. Bayley [3] during cooling tests with a water thermosiphon cycle. While analyzing the rate of filling of the channels isolated from one another, these authors observed -- by chance -- that the best performances recorded with a test-piece device rotating in a hot gas flow and cooled at their base by a flow of cold air were obtained with a small amount of water, less than 2% of the channel capacity. Figure 1 shows the values of the rate of heat flow measured as a function of the rate of filling and of the duration of the tests; beyond 30 minutes, the heat flux absorbed proves to be independent of the rate of filling up to a minimum value in the vicinity of 1.5%.

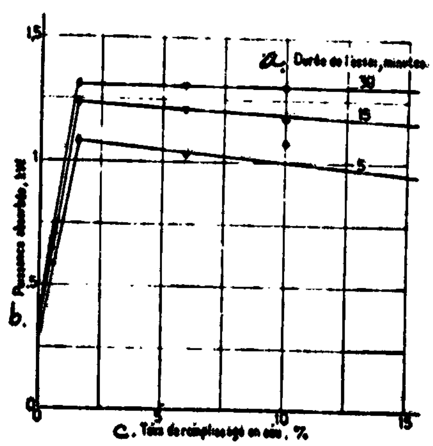

Fig. 1

Power absorbed by closed-cycle evaporation of water as a function of the rate of filling of the channels
Speed of rotation: 15 rpm^{-1} [3]

Key: a. Absorbed power; b. Duration of test, minutes; c. Rate of filling with water.

This lower limit of the fluid mass that should be introduced prior to sealing the channels is the result of the virtually uniform wetting condition of the wall exposed to the hot gases by the droplets condensed on the cooled wall, then centrifuged. The optimum rate of filling cannot be determined with accuracy in theory, and is determined only by experiment.

Condensation exerts a pumping effect in the centripetal direction, always provided that the radial gradient $dp/dr = \rho\omega^2 r$ (p = pressure, r = radius, ρ = density, ω = angular rotational velocity) does not attain excessive values, thus implying the use of a fluid with quite a low atomic mass. By way of indication, the ratio of pressures, with sodium, calculated by comparing the equation of state of vapor to that of the perfect gas, or $p/p_0 \approx \exp\left[\frac{\omega^2 m}{2kT}(r^2 - r_0^2)\right]$ will be close to 1.1 for a speed of rotation of 10^4 rpm^{-1}, a peripheral radius of $r = 0.3$ m, $r_0 = 0.1$ m and $T = 1156°K$ (normal boiling point of Na). Even if we take into account losses of pressure head suffered by the flow of the vapor in the channels traveled in the opposite direction by the droplets, a sufficient mass transfer is provided by the condensation process.

The recycled flow-mass in the steady state is fixed by the rate of heat exchange at the base, which is assumed to be cooled by the circulation of air through the rotor disk; this rate of transfer controls the level of the virtually uniform temperature at which the active portion of the blade is fixed.

3. Thermokinetic Characteristics of the Cycle with Sodium Phase Change /HE2.2.3

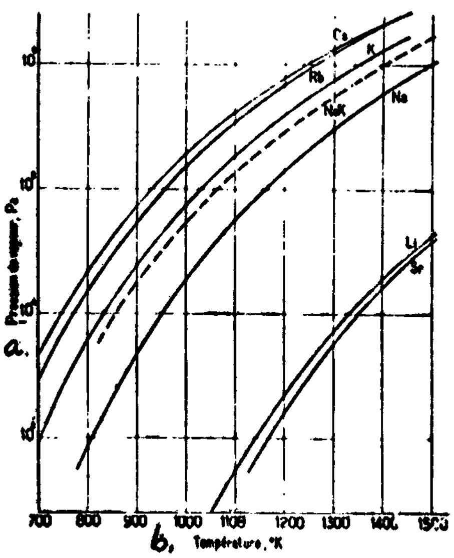

Fig. 2

Alkaline metal vapor pressure

Key: a. Vapor pressure; b. Temperature

Among the different metal whose vapor-pressure curves are shown in figure 2, sodium has the desirable temperature and saturation pressure characteristics for application to gas turbines. Vaporization heat, heat conductivity, viscosity, and surface tension of the liquid phase are favorable, and the melting point (98°C) is low enough to prevent any solidification under operating conditions.

Although the amount of heat absorbed by the blade, and therefore the deviation between the temperatures of its outer surface and of the hot gases are ultimately limited by the transfer capacity at the base acting as exchanger, it is important to provide for a channel geometry capable of ensuring quite a uniform distribution of the droplets and a suitable equality of temperature at the blades. An example of the configuration meeting this condition is shown in figure 3. The helical grooves of the inner wall of the channels and their pitch provide, in principle for the guidance of the droplets involved in the radial direction and for the desired uniformity of distribution.

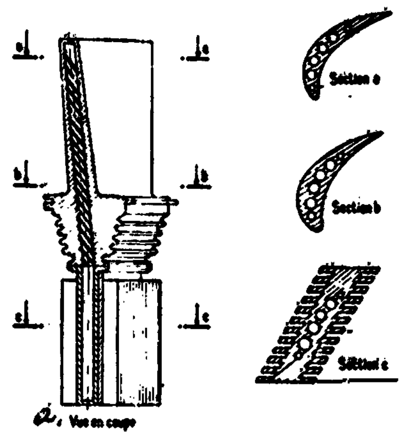

Fig. 3

Configuration of blades cooled by evaporation of a liquid metal

Key: a. Sectional view

Insofar as the ~~flow of~~ heat transmitted by the combustion gases to the blades of an aircraft engine turbine is concerned, when gas and blade temperatures are close to 1400 and 1150°K, the mean exhcange rate is of the order ot $\alpha_o \simeq 800$ W $m^{-2}(°C)^{-1}$ respectively [4]. Taking into account the dimensions of the blades, the flux Q_o absorbed by each of them is of the order of 2 kW, although the flux-mass of sodium vaporized at the normal boiling temperature (1156°K) would be equal to $Q_o/L_v = \dot{m}_v = 2000/3870 = 0.517$ gs^{-1} (L_v = specific vaporization heat).

With a cross section of the order of 1 cm^2 for the group of channels, the rate of vapor flow would this be:

$$u_v = \dot{m}_v/\rho_v A$$

or with $\rho_v \simeq 0.28$ kg m^{-3}, $u_v \simeq 18.5$ $msec^{-1}$, a very small value with respect to the speed of sound (a = 584 $msec^{-1}$ under conditions of thermodynamic equilibrium).

The result of this assessment is that the vapor flow takes place at too low a speed for the interaction with the droplets circulating against the current to be solely a function of the velocity imparted to these droplets by centrifugation; the great disparity in densities of the saturating vapor ($4\ m^3\ kg^{-1}$) and of the liquid sodium ($1.35 \cdot 10^{-3}\ m^3\ kg^{-1}$) leads to anticipate that this interaction is negligible in actual practice. A schematic analysis of the conditions of formation and separation of the droplets makes it possible, however, to determine this point.

Condensation enters in contact with the wall kept at a temperature lower than the local saturation temperature of vapor in the form of separate droplets. From discreet condensation nuclei the sodium deposited as moisture constitutes microbeads whose shape resembles truncated spheres, and whose adhesion is secured, during the initial phases of their growth by the surface tension forces to which the pressure of the vapor flux is added.. Under the conditions of intense artificial gravity occurring with centrifugal accelerations of the order of 10^4 to 10^5 g, the force of inertia plays a predominant role opposite bonding forces when the size of the beads becomes appreciable, although the formation of a continuous films from discreet condensation nucle appears to be excluded.

A simple computation makes it possible to determine the characteristics of separation of the beads; in the case where they are formed on an element with a surface assumed to be perpendicular to the radial direction of centrifugal acceleration, and where their adhesion is greatest, the equilibrium condition of a microbead is written (figure 4a):

$$(1+\cos\alpha)^2\left(1-\frac{1}{2}\cos\alpha\right)\Gamma R^2-\frac{3}{4}\frac{\rho_v}{\rho_l}u_v^2 K-\frac{3\sigma}{\rho_l}\sin^2\alpha=0 \qquad (1)$$

α designating the angle of contact, σ the liquid vapor surface tension, Γ centrifugal acceleration, R the radius of the truncated

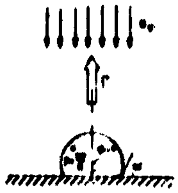

Fig. 4a

Forces acting on a bead in the process of fortion.

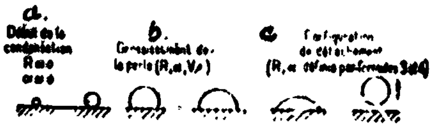

Fig. 4b

Successive shapes of a microbead to separation.

Key: a. Start of condensation;
b. Growth of bead;
c. Separation configuration (R, α, defined by formulas 3 & 4)

sphere, which the shape of the bead resembles, ρ_v and ρ_1 the densities of the vapor and liquid phases.

The volume of the bead, $V = \frac{2\pi}{3}(1+\cos\alpha)^2(1-\frac{1}{2}\cos\alpha)R^3$, must increase constantly due to the supply of mass maintained by vapor condensation; the separation of the bead must therefore enter when the volume defined by the value $R(\alpha)$ that satisfies the equilibrium equation (1) attains a maximum value. Beyond this value, the inertia term prevails over the dynamic pressure and surface tension terms, and the bead separates rapidly adopting the shape of and almost spherical droplet (figure 4b)

The extreme volume condition, or

$$\frac{1}{R}\frac{dR}{d\alpha} = -\frac{\sin^3\alpha}{(1+\cos\alpha)^2(2-\cos\alpha)} \qquad (2)$$

leads, following substitution of the value thus defined of $\frac{dR}{d\alpha}$ in the equation after derivation of (1) to equations:

$$R = -\frac{2\sigma}{\rho_v u_v^2}\,\frac{(1+\cos\alpha)(\cos^2\alpha+2\cos\alpha-1)}{(1-\cos\alpha)} \qquad (3)$$

$$\frac{8}{3}\frac{(\rho_v u_v^2/2)^2}{\sigma\rho_1\Gamma} = \frac{(1+\cos\alpha)^3(2-\cos\alpha)(\cos^2\alpha+2\cos\alpha-1)^2}{2(1-\cos\alpha)(3\cos^2\alpha-6\cos\alpha+1)} \qquad (4)$$

the second of which admits a solution only if the nondimensional parameter $\frac{8}{3}\frac{(\rho_v u_v^2/2)^2}{\sigma\rho_1\Gamma}$ is lower than a critical value in the neighborhood of $2.95\cdot10^{-3}$.

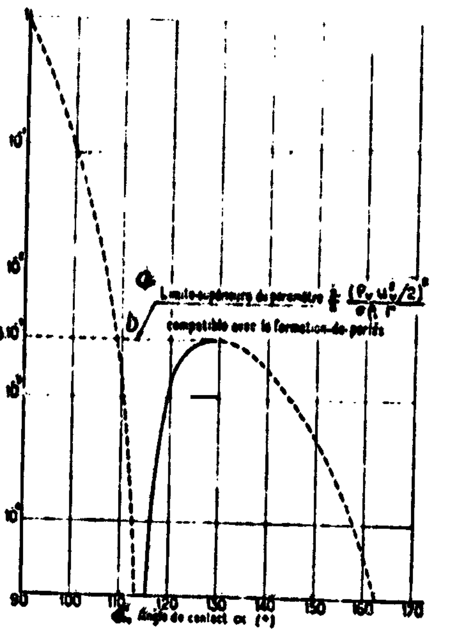

Fig. 5a

Characteristic function of the angle of contact of the condensed beads.

Key: a. Upper limit of the parameter ...
b. compatible with bead formation;
c. Angle of contact α (°).

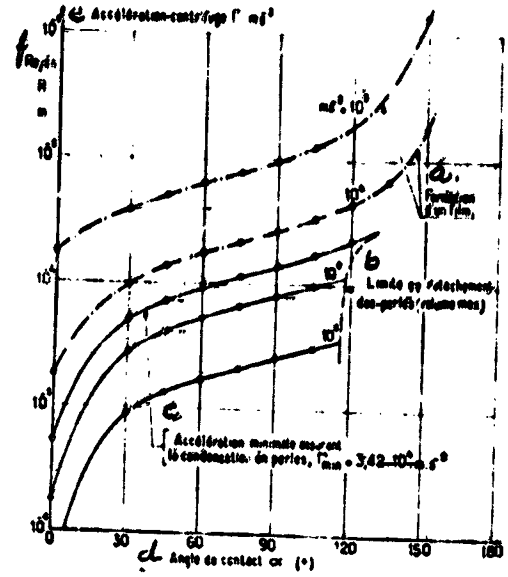

Fig. 5b

Change in bead radius, as a function of their angle of contact, according to the value of centrifugal acceleration (u_v = 25 msec^{-1}).

Key: a. Film formation;
b. Limit of bead separation (max. volume);
c. Minimal acceleration ensuring bead condensation;
d. Angle of contact; e. Centrifugal acceleration;
f. Radius.

In this case, the only solution in α that has a physical meaning falls between Arc cos $(1\sqrt{2}) \approx 114°$ and the value $\alpha_M \approx 129°$ corresponding to the maximum $2.95 \cdot 10^{-3}$ of the function that appears in the second member of (4). The other two apparent solutions, from the mathematical standpoint, correspond respectively to a negative value of R or to a minimum of V (figures 5a and 5b).

Under the previously retained conditions, ρ_v = 0.28 kg m^{-3}, ρ_1 = 740 kg m^{-3}, and depending on the value of σ calculated according to the correlation law proposed by Schonhorn [5] as a function of the difference in viscosities of liquid and vapor [6].

$$\sigma = 0{,}0?? \exp\left[- \frac{4{,}??.10^{-6}}{\mu_l - \mu_v}\right] \qquad Nm^{-1}$$

or $\rho \approx 0.154\ Nm^{-1}$, the separation criterion is written: /HE2.2.5

$$\Gamma \geq 5{,}8\ \frac{(\rho_v u_v^2/2)^2}{\sigma \rho_l} \qquad (5)$$

In the area of values of u_v, Γ where this inequality is satisfied, the radius R is itself lower than the valu R_M corresponding to the maximum admissible for the parameter $\frac{1}{2}\frac{(\rho_v u_v^2/2)^2}{\Gamma \rho_l \sigma}$, or according to (3) and with $\alpha_M \approx 129°$:

$$R_M \cong 0{,}149\left(\frac{\sigma}{\rho_l \Gamma}\right)^{\frac{1}{2}} \cong \frac{4{,}85.10^{-2}}{\Gamma^{1/2}}$$

For a given value of Γ, volume V remains virtually constant regardless of the value of u_v, and the radius R_g of the droplets formed after separation is thus close to the value

$$R_g = \left[\frac{(1+\cos\alpha_M)^2(2-\cos\alpha_M)}{4}\right]^{1/3} \cong 0{,}45\ R_M$$

whose evolution is shown in figure 6 as a function of Γ.

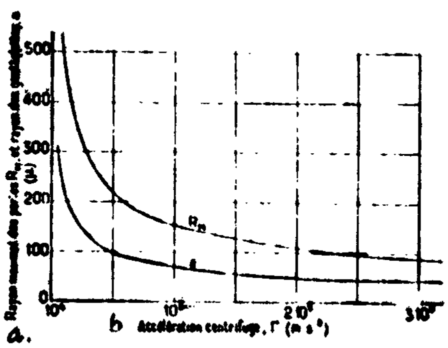

Fig. 6

Dimensions of beads and droplets, after separation.

Key: a. Maximum radius of beads, R_M, and radius of droplets, a;
b. Centrifugal acceleration.

Taking into account the order of magnitude for velocity u_v ($\leq 25\ msec^{-1}$), condensation would thus enter (with elements normal to the radius) in the form of a film in that portion of the rotor where acceleration would be less than $3.4 \cdot 10^4\ msec^{-2}$, i.e., in the case of an angular velocity of $\omega \approx 10^3\ sec^{-1}$, with a radius of less than $3.4 \cdot 10^{-2}$. The conditions achieved in practice very frequently preclude this contingency.

When condensation begins with a wall element this is oblique or parallel to the radial direction, the values of the radius R derived from a computation analogous to the preceding one would be evan lower, so that the aerodynamic resistance opposed to the movement or rolling of the microdrops must be estimated in laminar flow. Only the initial movement conditions differ according to the orientation of the wall element, since as the centrifugal force can only be imposed by the support driven at the rotational speed, the droplets assumed to be separated have a uniform movement of absolute velocity equal to that of their initial support. In related axes linked to the rotating frame, their trajectories are defined by the reduced equations

$$\begin{cases} s/r_0 = \mathrm{tg}\,\omega t - \omega t \\ r/r_0 = 1/\cos\omega t \end{cases}$$

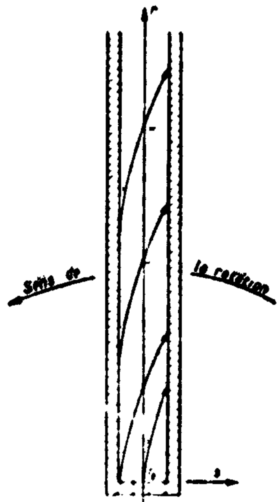

Fig. 7

Trajectories of droplets in relative motion in the middle plane of a cylindrical channel.

Key: a. Direction of rotation.

according to the notations in figure 7. The trajectories issuing from points situated at different values of the radius r_0 are homothetic, their shape being independent of angular velocity ω, which controls only the free passage of the droplets (the influence of the aerodynamic drag being neglected in a first approximation). The droplets are therefore rapidly intercepted by the wall towards the downstream side, i.e., towards the lower surfaces of the blades. They then roll towards the periphery, just as in the case of the beads condensed ab initio on this face. The droplets formed on the upstream side (situated on the upper surface) are subjected to the Coriolis acceleration from the moment their motion starts in the radial direction, which tends to separate them and project them downstream. In order to minimize this dissymetry affect in the distribution of the droplets, provisions must be made for a surface state capable of /HE2.2.6

guiding and slowing down the condensed particles, and of maintaining their adhesion.

In the limit, if we assume that friction is negligible, the motion of a drop rolling on the downstream wall would be described by the equation:

$$\frac{d^2r}{dt^2} = \omega^2 r$$

where $r = r_o \operatorname{ch} \omega t$; the complementary acceleration $2\omega \frac{dr}{dt} = 2\omega^2 (r^2 - r_o^2)^{1/2}$ comparable to the centrifugal acceleration would be expressed by a rapidly increasing pressure force tending to crush the drop against the wall.

In practice, the intensification of the heat tr[illegible]er r[illegible]t-ing from this pressure would no longer be unf[illegible], since it would act in a direction corresponding to tha[illegible] of the inequality of exchanges by external convection.

Insofar as taking the aerodynamic resistance to droplet motion into account, it would lead to writing the effective acceleration criterion as

$$C_x \frac{\rho_v u_v^2}{2} \pi R_g^2 \leqslant \omega^2 r \frac{4}{3} \pi R_g^3 \rho_l , \text{ ou } \frac{3}{8} \frac{\rho_v}{\rho_l} \frac{u_v^2}{\omega^2 r} C_x \leqslant R_g$$

C_x designating the drag coefficient, which is moreover only slightly different from the value $C_x = 12/Re$ of Stokes' law, given the low value of the Reynolds number $Re = \frac{\rho_v u_v R_g}{\mu_v}$.

Taking into account the change of C of the sphere as a function of Re [7], this inequality, which is written with $r = 0.1$ m, $u_v = 20 \text{ ms}^{-1}$:

$$R_g \geq 5.67 \cdot 10^{-7} \, C_x$$

actually leads to a minimum value of R_g, which is virtually equal to $7 \cdot 10^{-6}$ m for which $Re \simeq 1.1$ and $C_x \simeq 12.5$, conditions very close to Stokes' law.

4. Local Heat Exchange Conditions Due to Evaporation or Condensation

In an established flow, equal mass-flows of flui^ in the vapor state and in the liquid state circulate contercurrent in the channels. The power absorbed by vaporization is slightly greater than the power given up by condensation, due to conduction by walls that are not rig-rously isothermal and to variation of the change-of-state heat between extreme temperatures. It is important, in any event, to guarantee a effective wetting of the heating and cooling walls, the area of the moistened surfaces being a function of the values of the exchange coefficient related to boiling and condensation.

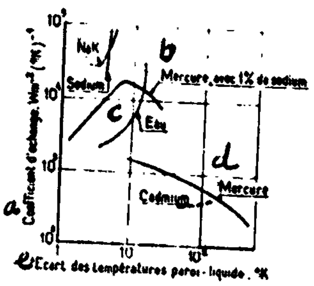

Fig. 8

Comparison of exchange coefficients by boiling of different fluids (according to [8].

Key: a. Exchange coefficient; b. Mercury with 1% sodium; c. Water; d. Mercury; e. Wall-liquid temperature variations.

By comparison with different metal fluids, Hg, Cd, NaK, as well as with water, the transfer coefficient measured boiling sodium are shown in figure 8 as a function of the variation in termperature between a stainless steel wall and the liquid [8]; sodium appears to be quite suitable for the transfer of high-density heat flux, which under the most favorable conditions, in nucleation regime and prior to the appearance of film boiling might reach 400 to 500 kW m^{-2}. /HE2.2.7

The data published by Katz [9] confirm the favorable effect of the wettability of alkaline metals, an effect that can be reinforced by the plating of the liquid metal by centrifugation, against any projecting surface element. It is therefore important to organize a suitable contour of the heating wall, whose useful surface ($\sim$ 50 cm^2) must permit absorption of the flux $Q_o \simeq 2$ kW previously retained according to a cited example. The absence of any incondensable element, the cleanliness of the surface state (preferably rough) of the wall are essential for transfer efficiency.

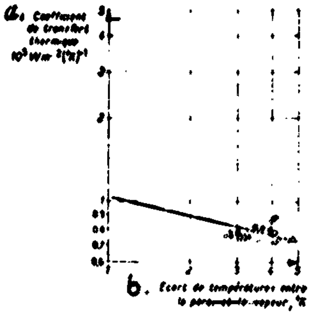

Fig. 9

Heat transfer coefficients measured by sodium condensation.

Key: a. Heat transfer coefficient; b. Temperature variations between wall and vapor.

The effect of centrifugation towards the cooled end, besides being less marked than at the periphery, is to facilitate condensation in microdroplets, a mode that was recognized by Misra and Bonilla [10] as suitable to provide for higher exchange coefficient values than in the case of film condensation. Although much lower than the theoretical forecast according to Nusselt's law, the heat flux values measured with sodium (figure 9) reach almost 400 kW m^{-2} under cooling conditions ensuring a temperature variation of 3 to 4°C between the wall and the liquid metal. The absence of incondensables, and cleanliness of the wall are still necessary to restrict the temperature drop at the solid-liquid interface, this drop would seem to explain, at least in part, the deviations noted with respect to the theoretical exchange coefficients.

In short, it seems possible to develop sufficient wet-surface areas, both on the active side of the blades where the fluid vaporizes and at their base that plays the role of condenser, to absorb a power of the order of 2 to 3 kW per blade.

It will be realized that it is still necessary to remove the heat flux supplied by condensation of the primary fluid by means of a secondary convection flow around the bases. This cooling, provided by the circulation of air drawn from the compressor in turbine-engine applications, can provide it thanks to different devices on the experimental study set-ups.

5. Experimental Study of the Process if Heat Transfer through Phase Change

Two set-ups were used successively in order first to verify the validity of the transfer principle with phase change under centrifugation, then to study the efficiency of this method of transfer under test conditions quite similar to those of the proposed application.

Firstly, tests at a moderate rotational speed were conducted using a revolving arm installed at ONERA. This device, which provided an acceleration of 10^3 msec^{-2} at about 1 m from the axis of rotation, serves as the support for a cylindrical chamber containing a small amount of ethyl acohol (from 10 to 20% of the total capacity) and vacuum-sealed at low temperature.

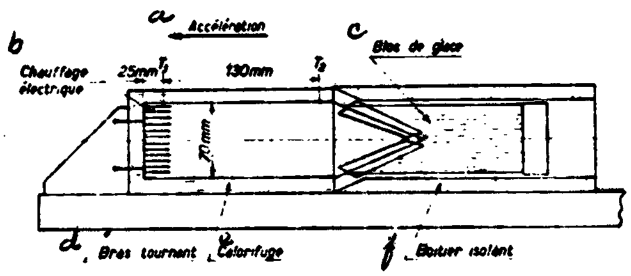

Fig. 10

Key: a. Acceleration; b. Electric heating; c. Block of ice; d. Revolving arm; e. Heat insulation; f. Insulating case.

According to the diagram shown in figure 10, a heating unit permits electrical adjustment of heat supply to the peripheral end, while the conical bottom equipped with blades is embedded in a block of ice contained in an insulating case, which acts as a heat sink. The centrifugal force constantly applies the ice against the bottom of the chamber, and the molten mass is measured as a function of the displacement of the block, recorded during the test.

The difference between temperatures T_1 and T_2 measured by means of thermocouples is a few degrees, so that the heat flux transferred by wall conduction appears very low with respect to the flux transferred by alcohol phase change. The pressure measured in the chamber is close to vapor pressure corresponding to temperature T_2. /HE2.2.8

The experiment with the revolving arms must be resumed at a higher temperature level ($\sim$ 600°C) with cesium as the transfer fluid. It is faced, however, with the drawback of a fairly substantial loss of power due to the effect of ambient air convection; the yield defined as the ratio of transferred power to electric power does not actually exceed 50% in the alcohol tests.

The test means used in the second stage of the study were carried out in 1966 by the Atomic Division of SNECMA, for the purpose of studying aircraft turbine blade cooling through the closed thermosiphon cycle [1].

This installation comprises essentially a rotor driven by a variable-speed electric motor (up to 8500 rpm^{-1}), with the peripheral acceleration reaching about $3 \cdot 10^5$ $msec^{-2}$. Two parallelepipedal test pieces representing two diametrically opposed blades are attached to the hollow disk and traversed by a flow of air that serves to cool the bases that act as exchangers (figure 11). They are perforated internally by five cylindrical channels with a 4-mm diameter partially filled with sodium and vacuum sealed.

The revolving unit is enclosed in a vacuum chamber; heat is supplied by a graphite-plate annular furnace where one can be heated by the Joule effect to a temperature of 2500°K, while the other acts as a reflector. A heat flux with a virtually uniform density is transferred to the test pieces. This furnace is insulated by screens and a layer of graphite wool, all of which is contained in a casing

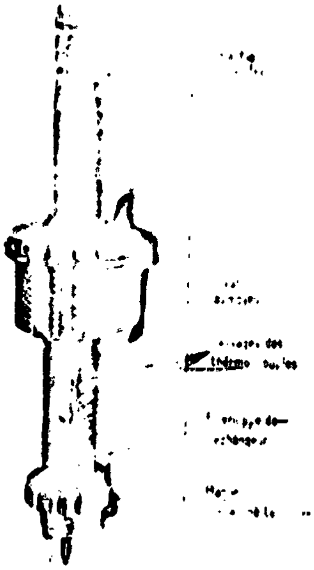

Fig. 11

Test piece on a high-speed rotor produced by the Atomic Division of S.N.E.C.M.A.

Key: a. Heated part; b. Attachment to hub; c. Thermocouple passages; d. Exchanger casing; e. Sealing ring.

with walls cooled by circulating water. In this manner, the heat balance can be established by cross-checking the external losses, the power radiated to the test pieces and to the hub; the power transferred through phase change is measured by the power collected by the base-cooling air, close to conduction losses through the shaft and the hub.

The test program in progress is based on the study of the effect of the sodium filling rate at different rates of rotational speed. The net transfer power through phase change will be determined by comparing it with the results obtained in the absence of sodium, with dry blades.

6. Conclusion

The application of an alkaline metal phase change process to the cooling of turbine engine blades has several advantages in principle; on the one hand, the limited space available inside the channels, which are necessarily very narrow, that can be arranged inside the blades is nevertheless sufficient to hold the minimum mass of liquid metal necessary to maintain the heat cycle; on the other hand, the quasi-constant temperature at which the active parts of the blades are kept facilitates heat exchange towards the bases.

The cooling itself is thus reported at the level of the exchangers, at the cost of an increased consumption of air taken from the compressor.

But it is only on the basis of the results of the experimental studies currently in progress that it will be possible to reach final conclusions regarding the favorable aspects and the limitations of using this device.

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